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BOUNDARY LAYER CONTROL AS A METHOD  
OF GAS TURBINE BLADE COOLING

A THESIS

Submitted to the Graduate Faculty  
of the  
University of Minnesota

by  
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for the  
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in  
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TABLE 1. INCREASED SOCIAL TRANSFER  
DURING THE PERIOD 1960-70

TABLE 1

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TABLE 1. INCREASED SOCIAL TRANSFER

TABLE 1

TABLE 1

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## OBJECT AND SCOPE

The object of this thesis was to determine the feasibility of cooling gas turbine blades by introduction of a controlled boundary layer of cool air over the blade surface.

This investigation included a static test of a single instrumented turbine blade in a variable high velocity, high temperature gas stream with variable cooling air flow. Two configurations of the test blade were used to produce variation in boundary layer control.

# APPENDIX

The object of this study was to determine the feasibility of making gas turbine blades by injection of a controlled boundary layer of air over the blade surface.

This investigation included a study of a single laminated injection blade in a variable high velocity, high temperature gas stream with certain modifications. The modification of the test blade was used to produce results in boundary layer control.

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## INTRODUCTION

Maximum effort in the development of gas turbines is being exerted to improve specific power output, to reduce specific fuel consumption and to increase reliability. The most promising field for the attainment of these objectives lies in increasing the turbine inlet temperature which is presently limited by permissible operating temperatures of blading materials. An investigation of the gas turbine thermodynamic cycle reveals the magnitude of improvement possible by increasing turbine operating temperatures. Such an investigation conducted by the NACA (Ref. 1) shows that for a given mass flow of working fluid the specific power output is proportional and the specific fuel consumption is inversely proportional to the inlet temperature. Fig. 1 illustrates this relation.

The increase of turbine inlet temperature, however, is limited by high temperature strength of blade materials. The development of high temperature metals is proceeding, but at a slow rate. How slowly metallurgical progress has been made is shown in Fig. 83 of Ref. 2. Allowable blade temperatures advanced from 1180° F. in 1935 to 1200° F. by 1940 and to 1385° F. by 1945. The rate of increase has been no greater since 1945.

Non-metallic materials such as ceramics have yet to demonstrate their adaptability to the rigorous service







requirements of turbine blading. As a result the use of some method of cooling the gas turbine blading presents itself as the method of allowing higher gas temperatures with present materials.

Several methods of blade cooling have been proposed and evaluated. A discussion of these methods as related to this thesis follows.

Late model German turbojet engines such as the Juno 004 employed hollow turbine blade cooled internally by means of air blown into the root and exhausted at the tip. 1650° F. turbine inlet temperature was used with 7% of compressor air output required to cool blades approximately 400° F.

The NACA has proposed (Ref. 1) an improvement to this method by inserting a core in the blade, leaving a small annular air passage. It was found that the heat transfer from blade to cooling air was principally in the boundary layer and adjacent cooling air so the insert permitted similar cooling with less air flow. Fig. 2 graphs the results of this improvement.

Another cooling method consists of circulating water through internal passages in the blade. This system of cooling has accomplished very large blade temperature reductions. German applications (Ref. 3) conducted by Dr. Schmidt permitted 850-930° F. blade temperatures with a gas temperature of 2200° F. Fig. 3 shows an NACA analytical

regulation of engine speed, as a result the use of  
 some method of cooling the gas turbine during operation is  
 well as the method of allowing slight gas temperature rise  
 during operation.

Several methods of blade cooling have been pro-  
 posed and evaluated. A discussion of these methods is pre-  
 sented in this paper.

The first method proposed consists of the  
 use of a cooling air duct which carries air from the  
 compressor to the turbine inlet. This air is then  
 cooled by the blades and is then used for cooling the  
 turbine inlet. This method is not very efficient  
 because the air is cooled by the blades and is then  
 used for cooling the turbine inlet.

The second method proposed (Ref. 1) is to  
 use a cooling air duct which carries air from the  
 compressor to the turbine inlet. This air is then  
 cooled by the blades and is then used for cooling the  
 turbine inlet. This method is not very efficient  
 because the air is cooled by the blades and is then  
 used for cooling the turbine inlet.

The third method proposed (Ref. 2) is to  
 use a cooling air duct which carries air from the  
 compressor to the turbine inlet. This air is then  
 cooled by the blades and is then used for cooling the  
 turbine inlet. This method is not very efficient  
 because the air is cooled by the blades and is then  
 used for cooling the turbine inlet.



investigation of water cooling which also gave considerable blade temperature reduction. It must be pointed out that while water cooling is very effective, the problems of handling the high temperature, high pressure water flow at high rates makes service application of this method difficult.

A modification of the foregoing system, known as rim cooling, has also been tried. Here water is circulated through the rotor rim so as to extract heat from the blade root. Less temperature reduction is obtained and the disadvantages of the water coolant system still exist. Fig. 4 shows rim cooling effectiveness. It can be seen from the figure that having a blade material of high thermal conductivity,  $K_M$ , is essential to this method.

All of the foregoing methods employ the same basic principle of cooling. They do not inhibit the heat transmission to the blade, but do increase the internal conductivity, or removal of heat, thereby affecting cooling. The proposal of this thesis is to substitute a boundary layer of cool air over the blade's surface in order to inhibit the heat transmission from the hot gases to the blade.

A coating of high temperature ceramic of low conductivity would embody this same principle. Fig. 5 shows the effectiveness of ceramic coatings of various thicknesses. While this is a very promising field insofar as temperature of operation is concerned, the inherent defects of brittle-

Investigation of water cooling which also gave considerable  
 of the comparative reduction. It must be pointed out that  
 while water cooling is very effective, the results of  
 feeding the high temperature, high pressure water from a  
 high water source involve attention at this point also.  
 Well.

A modification of the foregoing system, known as  
 the cooling, has also been tried. Here water is circulated  
 through the tower and is so heated that from the glass  
 front. Such temperature reduction is obtained and the dis-  
 advantages of the water cooling system will be noted. It  
 shows the cooling effectiveness. It can be seen from the  
 figure that using a single system of high thermal capacity  
 water, it is possible to do this work.

All of the foregoing methods employ the same  
 basic principle of cooling. They do not include the heat  
 transmission in the liquid, but do require the internal  
 conductivity, or transfer of heat, thereby affecting cooling.  
 The purpose of this device is to establish a boundary  
 layer of cool air over the liquid's surface in order to re-  
 duce the heat transmission from the hot gases to the liquid.  
 A cooling of high temperature liquids of low vis-

cosity will result. This same principle will be shown  
 the effectiveness of various systems of various substances.  
 While this is a very practical field in the laboratory  
 of operation is concerned, the present status of this



ness, thermal shock sensitivity and low tensile strength have obviated service use of ceramic covered blading.

If a region of low thermal conductivity can be interposed between the gas and the blade, then the objective of blade cooling could be accomplished. The natural boundary layer on the blade is such a region. However, the natural boundary layer forms at the temperatures of the gas. In this experiment the use of relatively cool air from the engine compressor is suggested to form a lower temperature boundary layer.

The justification of this idea is based on one of the fundamental laws of heat transfer, Faurier's equation for conduction (Ref. 4). (Experience has shown radiation effects to be secondary). Stated mathematically for steady state conduction:

$$q = KA \frac{dt}{dx}$$

where  $q$  = rate of heat transfer.

$K$  = coefficient of thermal conductivity.

$A$  = crosssectional area of path.

$\frac{dt}{dx}$  = temperature gradient in direction of heat flow per unit distance.

This law shows that for a given configuration the rate of heat transfer from gas to blade may be made by reducing  $K$  and/or  $\frac{dt}{dx}$ .

$K$  for air is reduced by reducing temperature.

This is shown mathematically from Lucheng equation:

heat, thermal conductivity and low thermal capacity

have obtained results on the various physical properties.

If a system of low thermal conductivity can be

interposed between the gas and the blade, then the objective

of blade cooling could be accomplished. The natural tendency

of any layer on the blade is to form a porous structure, and

natural porous layer forms at the temperature of the gas.

In this experiment the use of relatively small air flow the

system component is suggested as being a most important

feature of the

The construction of this test is based on the use of

the fundamental laws of heat transfer, Fourier's equation

for conduction (Eq. 1), (convection Eq. 2) and radiation

applied to the geometry. (Detailed mathematically for steady

state conditions)

$$Q = \frac{kA \Delta T}{L}$$

where  $Q$  = rate of heat transfer.

$k$  = coefficient of thermal conductivity.

$A$  = cross-sectional area of path.

$L$  = thickness of material in direction of heat

flow (see Fig. 1).

This law shows that for a given configuration the

rate of heat transfer from gas to blade may be made by re-

$$Q = hA(T_g - T_b)$$

where  $h$  is the heat transfer coefficient.

This is shown mathematically from Fourier's equation

$$K = K_{32} \frac{492 + C}{T + C} \left( \frac{T}{492} \right)^{3/2}$$

where  $T$  = absolute temperature

$C$  = constant (.0129 for air).

$K_{32}$  =  $K$  at 32° F.

The temperature gradient from the boundary layer to blade,  $\frac{dt}{dx}$ , is reduced by the use of the cool air controlled boundary layer. In fact, the cool air boundary layer will at first be lower in temperature than the blade so that the blade will transfer heat to the boundary layer. However, the temperature gradient from the hot gas to the boundary layer would be increased so it would be rapidly heated. The optimum configuration might therefore require a series of bleeds from the blade so the average temperature of the layer along the blade would be minimized.

In the author's experience a controlled boundary layer has been successfully employed to cool a liquid rocket nozzle. In 1936 the author collaborated in the construction of a liquid rocket motor in which a boundary layer of coolant air bled into the nozzle enabled prolonged operation. The nozzle was of mild steel yet endured the very high temperature rocket exhaust gases better than any contemporary nozzles of superior materials.



1918 and 1919, 2 specimens \* 2

The temperature gradient from the boundary layer to plate,  $T_b$ , is reduced by the use of the heat sink design. In fact, the heat sink boundary layer will at first be lower in temperature than the plate as heat the plate will transfer heat to the boundary layer. However, the temperature gradient from the hot gas to the boundary layer will be increased as it would be without fins. The optimum configuration might involve having a series of plates from the plate to the average temperature of the gas along the plate walls or channels.

in the machine's operation a similar condition  
never has been successfully repeated to such a degree  
before. In 1955 the engine exhibited in the com-  
parison of a light engine test in which a steady  
layer of coolant air did not the same stable pressure  
operation. The reason for this was not known at  
very high temperatures under which the engine  
operated under a similar condition.



## TEST EQUIPMENT

Fig. 5 shows the complete test equipment layout schematically. The test blade was mounted in a closed test section supplied with hot gas from a single J-33 combustion chamber. Fig. 6 is a photograph of the test section mounted on the burner. Air was supplied to the burner from the compressor of a naturally aspirated Allison V-1710 engine, Fig. 7. The quantity of gas flow was regulated by throttling the Allison engine while its temperature was controlled by burner fuel pressure. The temperature of the blade was measured by two thermocouples. All control and measurement was done from the control panel adjacent to the gas turbine test cell. Fig. 8 is a photograph of the control panel. The air which formed the controlled boundary layer was supplied from the laboratory air main at regulated pressure. The quantity of cooling air was measured in a standard design sharp edged orifice meter, shown in Fig. 9.

The test blade was manufactured from a solid Juno 004 turbine blade. Availability was the reason for selection of this blade. The "tinidur" type alloy (30% nickel, 14% chrome, 1.75% titanium, 12% carbon, balance, iron) possessed very difficult machining properties and low thermal conductivity. The blade roots were cut off flat for convenient mounting and the tip shortened by 3/4 inch

# TEST EQUIPMENT

Fig. 3 shows the complete test equipment layout.

essentially. The test stand was mounted in a closed steel  
enclosure supplied with air from a single 1-1/2 inch  
diameter. Fig. 4 is a photograph of the test stand enclosure.

on the burner. Air was supplied to the burner from the  
compressor of a naturally aspirated piston 7-1/2 inch,  
Fig. 5. The quantity of air flow was regulated by means

with the piston engine valve for temperature was controlled  
by means of a piston. The temperature of the piston was  
controlled by two thermocouples. All control and measurement

was done from the control panel adjacent to the gas turbine  
test cell. Fig. 6 is a photograph of the control panel.  
The air which formed the controlled boundary layer was supplied

from the laboratory air main at regulated pressure.  
The quantity of heating air was measured in a standard 1-1/2  
inch square orifice water meter, shown in Fig. 7.

The test stand was constructed from a single  
large steel section plate. Availability was the reason for  
selection of this plate. The "standard" type alloy steel

alloy, 1/2 inch, 1.75 inch, 1.5 inch, 1.25 inch, 1.0 inch, 0.75 inch,  
from processed very difficult working properties and the  
standard construction. The plate was cut off the

for convenient mounting and the air delivered by 1/2 inch



because of space limitations in the test section.

For the first test, configuration A was manufactured. In this blade the cooling air supply hole was drilled up the blade from root to  $1/4$  inch of tip through the thickest section. This hole was .20 inch in diameter. The air bleed holes ( $1/16$  inch) were drilled from blade surfaces joining the supply hole. They were placed at a 45 degree angle with blade surface. There were six bleed holes to each surface. The exits were ground out with a fish tail countersink pattern to distribute the bleed air spanwise. A .15 inch hole was drilled up the leading edge for location of the thermocouple tip at midspan. The trailing edge was too thin to permit similar treatment so a  $3/32$  inch hole was drilled chordwise at midspan to snugly hold a thermocouple bead. The thermocouple was lead in in a stainless steel tube one inch downstream and bent 90 degrees and cemented in the trailing edge hole. Figs. 12, 13 and 14 show the thermocouple mounting. Fig. 10 shows the A blade between a standard blade and a shortened standard blade. The boundary layer is introduced at approximately the 30% chord point.

Configuration B blade is shown in Fig. 11. The boundary layer air supply was introduced through a .15 inch drilled passage  $1/4$  inch behind the leading edge. A 60 degree included angle slot was milled down the length of the leading edge and  $3/32$  inch bleed holes drilled joining the

because of minor limitations in the test results.

For the first test, configuration 1 was considered.

Based on this study the resulting air supply rate was  
 calculated by the blade from test to 1/2 inch at 1/2 inch.  
 The highest velocity. This rate was .25 inch in diameter.  
 The air flow rate (1/2 inch) was drilled from 1/2 inch  
 pressure against the supply rate. They were drilled at a  
 1/2 degree angle with 1/2 inch diameter. These were 1/2 inch  
 holes in each surface. The rate was found to be a  
 1/2 inch full compressing system to distribute the 1/2 inch air  
 quantity. A 1/2 inch hole was drilled up the leading edge  
 for injection of the thermocouple tip at midpoint. The 1/2 inch  
 hole was 1/2 inch in diameter and 1/2 inch in length. The 1/2 inch  
 hole was drilled through at 1/2 inch to 1/2 inch.  
 A thermocouple hole. The thermocouple was fixed in a  
 stainless steel tube with 1/2 inch diameter and 1/2 inch  
 length. The hole was in the leading edge hole. 1/2 inch 1/2  
 inch 1/2 inch and thermocouple mounting. 1/2 inch 1/2 inch  
 the 1/2 inch between a standard 1/2 inch and a standard 1/2 inch  
 air hole. The boundary layer is indicated as follows:  
 mainly the 1/2 inch hole.

Configuration 2 study is shown in fig. 11. The  
 boundary layer air supply was indicated through a 1/2 inch  
 drilled hole. 1/2 inch in the leading edge. A 1/2 inch  
 hole was drilled into the leading edge from the 1/2 inch at the  
 leading edge and 1/2 inch hole was drilled from the 1/2 inch at the



supply passage. In this design the cooling enters the slot opposed by stagnation pressure and flows out of the slot on both edges, forming the boundary layer.

The test section consisted of the blade mounting block shown in Fig. 13, and two side plates made of six inch channel. The bottom was closed with a 1/2 inch plate so that the hot gases which entered at the top were constrained to exhaust through the open side. The entrance and exit dimension are 3.5 x 4.5 inches. Fig. 12 pictures the complete test section. Fig. 14 shows another view of the blade mounting block. The test blade is centrally located with two parallel mounted standard blades to guide the flow.

Temperatures of the test blade were measured by 20 gauge chromel-alumel thermocouples which read on a Brown recorder. The small size thermocouples provided fast response and use of standard sillimanite insulations in the blade. The insulators were ground slightly in diameter for mounting in blade. The thermocouple tips were firmly seated in 3/32 inch holes for maximum sensitivity to blade temperatures. The trailing edge thermocouple bead was buried completely to insure it would sense blade, rather than gas temperatures.

Compensated lead wires connected the thermocouples to the selector switch for the recorder to eliminate errors in readings by variation in ambient temperature. The gas

supply channels. In this design the venting system for the  
 exposed by atmospheric pressure and lines out of the area in  
 both edges, located the secondary layer.

The test section consisted of the blade mounting

blades shown in Fig. 1, and the blade design was of the  
 last design. The bottom was closed with a 1/2 inch plate  
 so that the hot gases were trapped at the top and con-  
 sidered as exhaust through the open side. The exhaust  
 and exit diameter was 5.0 x 4.0 inches. The 1/2 inch  
 the complete test section. The 1/2 inch section was of  
 the blade mounting block. The test blade is completely iso-  
 lated with the previous mounted standard blades to reduce the  
 flow.

The purpose of the test blade was designed to  
 be a pure thermodynamic thermometer which would be a pure  
 resistor. The test blade thermodynamic standard test re-  
 sistor was not a standard thermodynamic resistor in the  
 blade. The thermodynamic were placed slightly in diameter for  
 mounting in blade. The thermodynamic also were fixed in  
 in 1/2 inch holes for uniform sensitivity to blade design.  
 edges. The existing edge thermodynamic had not tested con-  
 siderably to insure it would make blade, tested from the

thermodynamic.  
 The thermodynamic test after mounted the thermodynamic  
 to the resistor which for the resistor in standard form  
 in testing of resistor in standard resistor. The test

temperature entering the test section,  $T_4$ , was measured by a radiation shielded chromel-alumel thermocouple.

temperature between the two points, 11.5 and 12.5, was 0.5 degrees.  
 a radiation shielded against the sun.

There is a small amount of water in the soil.

The soil is very dry and the water is very low.

The soil is very dry and the water is very low.

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The soil is very dry and the water is very low.



## TEST PROCEDURE

The temperature of the test blade was read with and without cooling air flow under exactly similar flow conditions. This technique permitted comparison of the two temperatures obtained to show the blade reduction due to cooling air alone.

The Allison engine was first started and its speed set to obtain the desired flow rate of burner air. The flow rate was measured by means of the pressure drop across the orifice in the compressor inlet duct.

Next, combustion was initiated in the burner with the spark and acetylene flare and fuel pressure adjusted until desired gas temperature obtained. When conditions stabilized temperatures and pressures were recorded. During runs with cooling air, the flow rate was varied in increments of 1/10 inch of water and temperature recorded when stabilized.

In order to investigate the cooling effects over a broad range of gas velocities three settings of the Allison engine were used to give low, medium and high gas flow rates. However, the low flow rate is not included in this report as it was unrealistically low compared with actual turbine operation. The flow rate was below minimum idling rate for a J-33 engine.

# TEST RESULTS

The temperature of the test fluid was read with

and slight cooling air flow under exactly similar flow conditions. This temperature provided comparison of the test results obtained to show the same position was in cooling air alone.

The test results were first stated and the speed was adjusted to obtain the desired flow rate of burner air. The flow rate was measured by means of the pressure drop under the orifice in the combustion inlet duct.

Next, combustion was initiated in the burner with the speed and cooling air flow and fuel pressure adjusted until desired gas temperature was reached. Then readings of gas temperature and pressure were recorded. During each cooling air flow rate was varied in the amount of 1/10 inch of water and temperature recorded when stabilized.

In order to investigate the cooling effect of a small amount of gas velocity the same amount of the cooling air flow was used as first test, cooling and with gas flow rate. However, the low flow rate is not included in this report as it was unrepresentative of the test conditions. The flow rate was varied in the amount of 1/10 inch of water and temperature recorded when stabilized.

## TEST RESULTS

The results of the experiment are contained in tables I and II and the graphs, Figs. 16 through 19. The graphs are plotted to show temperature reduction versus weight of cooling air flow.

These graphs are similar in shape and show that the reduction in blade temperature was approximately twice as great in the leading edge as the trailing edge. This is to be expected because of the increase in boundary layer temperature resulting from heat transmission from the gas. Also, the thinness of the blade section near the trailing edge offers more resistance to heat flow internally.

The graphs also show that the temperature reduction rate is greatest (the slope is maximum) at low cooling air rates. This is evidence that the boundary layer is established at low flow rates and is effective in reducing heat transfer. Beyond a flow rate of .2 lb./min. most of the graphs become straight line functions. This apparently results from thickening of the boundary layer and shows the insulating effect is proportional to the thickness. This effect conforms with Fourier's law. The cooling effectiveness, particularly at low flow rates, is greater with this method than the method of Fig. 2.



# THE RESULTS

The results of the experiment are contained in  
Tables I and II and the graphs, Figs. 10 through 12. The  
graphs are plotted in three temperature reduction versus  
weight of cooling air flow.

These graphs are similar in shape and show that  
the reduction in flame temperature was appreciably lower  
at first in the burning stage as the cooling air flow  
is to be expected because of the increase in boundary layer  
temperature resulting from heat transmission from the gas.  
Also, the reduction of the flame temperature near the trailing  
edge of the nose resistance is less than initially.

The graphs also show that the temperature reduction  
rate is greatest (the slope is greatest) at the end  
of the test. This is evidence that the boundary layer  
is established at the first stage and is effective in re-  
ducing the temperature. Beyond a flow rate of 2.5 lb/min.  
most of the graphs show slightly less resistance. This  
apparently results from the reduction of the boundary layer  
and shows the increasing effect is proportional to the  
boundary. This effect is shown with the test results. The  
cooling effectiveness, particularly at low flow rates, is  
greater with this material than any material of Fig. 10.



To illustrate the cooling effectiveness consider the J-33 turbojet engine. Maximum cooling of 280 F. at 1600 F. gas temperature could be accomplished with only 2% of compressor air.

Configuration A produced more uniform results than those of configuration B, as can be seen by comparing Figs. 16 and 17 with 18 and 19. Configuration A curves plotted more parallel and gave results proportional to gas temperature, while configuration B curves intersect and are out of order with gas temperature increments. Configuration A probably gave more uniform boundary layer formation, since the flows were convergent rather than opposed. It was calculated that the stagnation point on the leading edge would fall in the milled slot so the cooling air would spill over both surfaces of the blade and form good boundary layers. From the non-uniform results at different flow rates stagnation point shifting may be indicated. Also, turbulence in the test section may have prevented uniform boundary layer formation and promoted mixing. It is believed that had the blade been manufactured with boundary layer control slots of the type used in airplane practice, much higher quality results would have been obtained. This type bleed was considered but discarded because of the machining problems, which would have required machining beyond shop capacity.

The illustration shows the cooling characteristics of the

the 1-25 inch diameter, 1000 ft. long, 1000 ft. diameter

1000 ft. long, 1000 ft. diameter, 1000 ft. long, 1000 ft. diameter

of the cooling air.

Consequently, the cooling air is not only

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Uncontrollable variation in the speed of the Allison engine contributed to inaccuracy of data. A "hunting" of about fifty RPM occurred during much of the running, which produced 500 RPM variations in compressor speed. The variation in burner air flow caused drifting of gas temperatures.

Comparison of Figs. 16 with 17, and 18 with 19, shows cooling effectiveness variation with gas flow rate. Cooling effectiveness is greater at the lower flow rate. This is consistent with the laws of heat transmission by convection. The heat transfer from gas to blade increases with velocity.

From all graphs, the blade temperature reduction is shown to increase with gas temperature. This, also, is compatible with the laws of heat transfer, the  $\frac{dt}{dx}$  term of Fourier's equation increases so more heat is transferred to the outer boundary layer. But, the heat transfer to the blade through the laminar sublayer is of such small magnitude that the net effect is greater blade temperature reduction.



temperature variation in the speed of the

Alison's speed variation is inversely of the  
"radius" of about 15% for speeds varying from 0 to  
1000, which produces 100% variation in temperature  
speed. The variation in speed is also caused by the  
of the temperature.

Temperature of the air is 15, and is with 10,  
above cooling atmospheric variation with the air.  
Cooling atmospheric is caused by the loss of heat  
this is consistent with the law of heat transmission by  
convection. The heat transfer from the air to the  
with velocity.

From all stages, the data indicates that  
is about 10 degrees with the temperature. This, also, is  
consistent with the law of heat transfer, the  $\frac{1}{2}$  law of  
Newton's law of cooling, which states that the rate of  
of the heat transfer is proportional to the difference in  
the fluid through the liquid surface is of such small  
magnitude that the net effect is greater than the  
temperature.

## CONCLUSIONS

In view of the limited scope of the experimental tests, no detailed quantitative conclusions can be drawn. However, the results obtained from the foregoing tests do support the following general conclusions:

1. The introduction of a boundary layer of relatively cool air on a turbine blade in a high temperature, high velocity gas stream inhibits the transmission of heat from the gas to the blade, more than through the natural boundary layer, and results in reduction of blade temperature.

2. The magnitude of the reduction in blade temperature is proportional to the weight flow of air introduced into the boundary layer up to the limit investigated of 2% of the gas flow in an equivalent full scale engine.

3. This method of blade cooling is feasible insofar as weight flow of cooling air required to accomplish useful blade temperature reduction is concerned.

# EXPERIMENTAL

In view of the limited scope of the experimental tests, an extended qualitative discussion can be drawn. However, the results obtained from the foregoing tests do support the following general conclusions:

1. The introduction of a secondary layer of this highly elastic air on a turbine blade in a high temperature, high velocity gas stream results in the formation of a thin film on the blade, which tends to reduce the surface temperature, and results in reduction of blade temperature.

2. The reduction of the temperature of the blade surface is proportional to the weight flow of air introduced. The secondary layer is the lightest layer of air introduced of all of the flow in the turbine. This means that the weight flow of air introduced is proportional to the weight flow of air introduced.

3. The weight of blade cooling is proportional to the weight flow of cooling air. The weight of cooling air is proportional to the weight flow of cooling air. The weight of cooling air is proportional to the weight flow of cooling air.



# OBSERVED TEST DATA

TABLE I

## HIGH AIR FLOW RUNS

COMPRESSOR RPM - 24,000

— GAS TEMPS —

COOLING AIR		800°F				1000°F				1200°F				1400°F				1600°F			
ΔP "H <sub>2</sub> O	W <sub>a</sub>	T <sub>BLE</sub>	T.R.	T <sub>STE</sub>	T.R.	T <sub>BLE</sub>	T.R.	T <sub>STE</sub>	T.R.	T <sub>BLE</sub>	T.R.	T <sub>STE</sub>	T.R.	T <sub>BLE</sub>	T.R.	T <sub>STE</sub>	T.R.	T <sub>BLE</sub>	T.R.	T <sub>STE</sub>	T.R.
0	0	785		785		995		995		1190		1195		1390		1390		1585		1585	
.1	.217	720	65	760	25	920	75	965	30	1110	80	1150	45	1255	135	1325	65	1440	145	1615	70
.2	.345	710	75	755	30	920	75	965	30	1070	120	1135	60	1255	135	1325	65	1440	145	1505	80
.3	.471	700	85	750	35	895	100	950	45	1050	140	1135	60	1220	170	1310	80	1385	200	1480	105
.4	.600	690	95	735	50	875	120	945	50	1030	160	1110	85	1200	140	1300	90	1350	235	1465	120
.5	.726	675	110	730	55	855	140	937	58	1015	175	1115	80	1185	205	1290	100	1335	250	1450	135
.6	.846	665	120	730	55	850	145	930	65	1015	175	1115	80	1160	230	1270	100	1320	265	1450	135

RUN 1

CONFIGURATION A

0	0	755		755		945		975		1150		1150		1325		1330		1480		1475	
.1	.217	725	30	750	5	915	30	965	10	1080	70	1120	30	1300	25	1318	12	1350	130	1440	35
.2	.345	695	60	735	20	900	45	955	20	1075	75	1120	30	1230	75	1310	20	1330	150	1440	35
.3	.471	670	85	720	35	875	75	945	30	1042	108	1120	30	1200	125	1310	20	1310	170	1445	50
.4	.600	650	105	720	35	855	90	935	40	1035	115	1120	30	1160	165	1295	35	1300	180	1425	50
.5	.726	640	115	715	40	825	120	920	55	1000	150	1115	35	1135	190	1290	40	1270	210	1420	55
.6	.846					800	145	905	70					1110	215	1275	55	1255	225	1420	55

RUN 2

CONFIGURATION B

		800°F	1000°F	1200°F	1400°F	1600°F
P <sub>f</sub>	PSI	67	76	84	96	107
W <sub>f</sub>	lb/hr	74	93	107	129	146
ΔP <sub>B.A.</sub>	"Hg	1.83	1.82	1.80	1.65	1.50
W <sub>B.A.</sub>	lb/sec	2.85	2.845	2.83	2.71	2.59
P <sub>3</sub>	"Hg	14.8	15.0	16.1	17.1	16.8
P <sub>E3</sub>	"Hg	15.4	15.4	16.5	17.5	17.2
P <sub>4</sub>	"Hg	.72	.64	.60	.52	.50
P <sub>E4</sub>	"Hg	11.45	11.8	13.1	13.75	13.8
g	"Hg	10.63	11.16	12.5	13.23	13.3
M <sub>4</sub>		.731	.75	.795	.82	.823

TEMP: COOLING AIR - 80°F  
 AMBIENT (TEST CELL) 120°F

PRESS: ATMOS - 29.82 "Hg  
 AMBIENT (TEST CELL) 27.50 "Hg



OBSERVED TEST DATA  
MEDIUM BURNER AIR FLOW RUNS  
COMPRESSOR RPM - 13,250

-16-

TABLE II

COOLING AIR		TEMP'S																CONFIGURATION A				
ΔP	"H <sub>2</sub> O	800°F				1000°F				1200°F				1400°F					1600°F			
W <sub>a</sub>		T <sub>BLE</sub>	T <sub>R</sub>	T <sub>STE</sub>	T <sub>R</sub>	T <sub>BLE</sub>	T <sub>R</sub>	T <sub>STE</sub>	T <sub>R</sub>	T <sub>BLE</sub>	T <sub>R</sub>	T <sub>STE</sub>	T <sub>R</sub>	T <sub>BLE</sub>	T <sub>R</sub>	T <sub>STE</sub>	T <sub>R</sub>		T <sub>BLE</sub>	T <sub>R</sub>	T <sub>STE</sub>	T <sub>R</sub>
0	0	777		785		995		1000		1115		1200		1310		1345		1535		1570		
.1	.217	720	57	750	35	912	95	763	37	1052	143	1135	65	1210	120	1338	51	1470	15	1550	40	
.2	.345	710	67	750	35	815	100	750	50	1042	150	1121	73	1230	100	1345	70	1422	103	1540	50	
.3	.471	645	82	750	35	860	135	740	60	1035	160	1120	80	1202	188	1312	83	1388	117	1550	40	
.4	.600	685	72	750	35	850	145	740	60	1015	180	1120	80	1175	215	1303	72	1365	220	1550	40	
.5	.726	610	107	750	35	835	160	740	60	1010	185	1120	80	1150	240	1300	75	1340	245	1550	40	
.6	.846	655	122	750	35	825	170	740	60	985	210	1120	80	1135	255	1300	75	1315	270	1540	50	
.7	.972	650	127	750	35	815	180	740	60	975	220	1120	80	1120	270	1300	75					
.8	1.09	645	132	750	35	800	175	730	70	965	230	1120	80									
0	0	720		725		730		745		1142		1147		1325		1320		1440		1480		
.1	.217	645	75	706	11	860	170	730	15	1042	100	1120	21	1188	137	1245	35	1385	125	1455	25	
.2	.345	640	80	700	25	850	130	725	20	1030	112	1120	21	1182	143	1235	35	1365	135	1455	25	
.3	.471	632	88	700	25	831	133	725	20	1015	127	1110	37	1175	150	1275	45	1336	164	1450	30	
.4	.600	625	75	700	25	825	205	725	20	1000	142	1100	47	1120	205	1270	50	1275	215	1438	42	
.5	.726	605	115	675	30	800	230	715	30	965	177	1075	52	1103	217	1255	67	1250	240	1425	55	
.6	.846																					
.7	.972																					
.8	1.09																					
INSUFFICIENT COMPRESSED AIR SUPPLY																						CONFIGURATION B

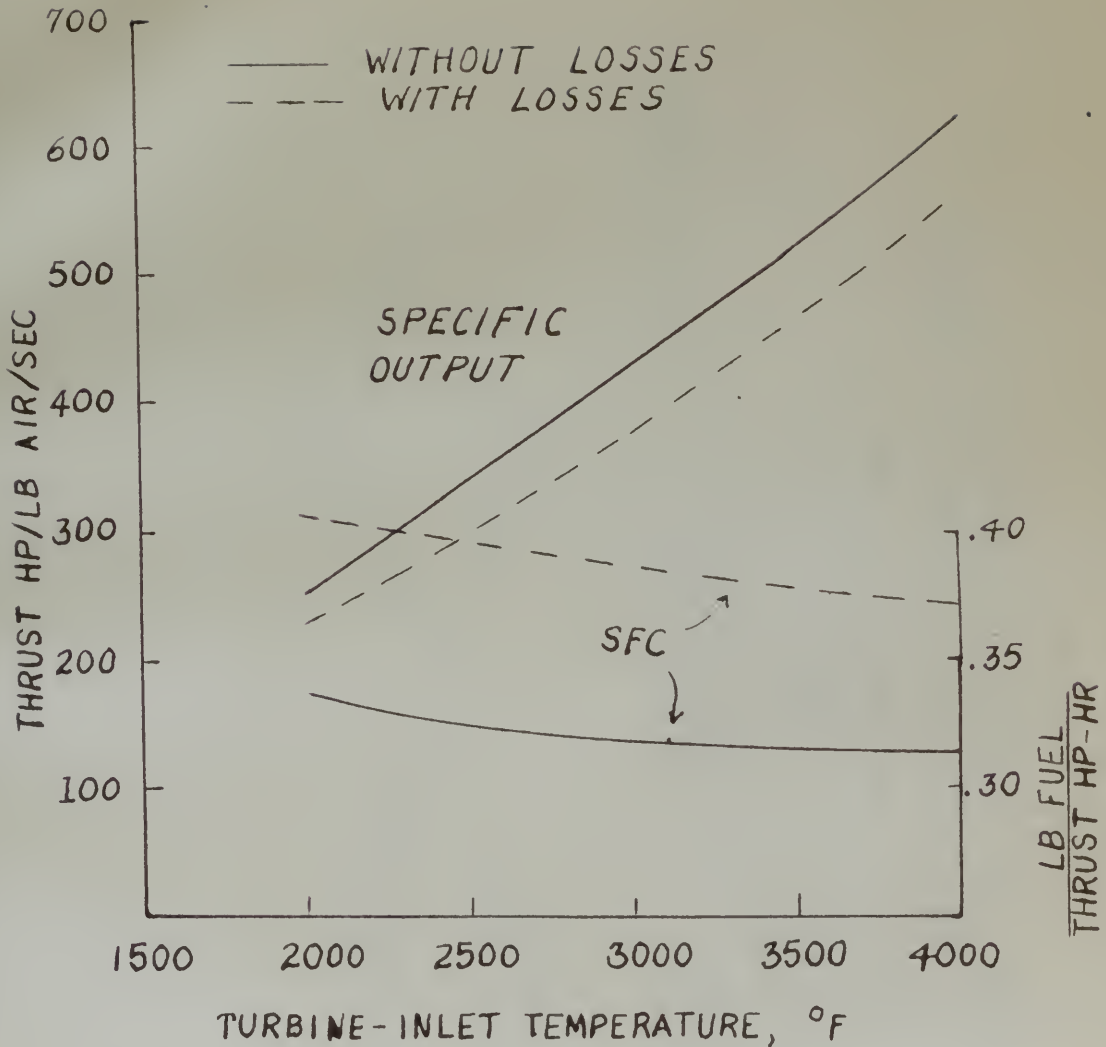
INSUFFICIENT COMPRESSED AIR SUPPLY

	800°F	1000°F	1200°F	1400°F	1600°F
P <sub>t</sub> PSI	47	61	67	73	79
W <sub>t</sub> lb/hr	47	65	80	90	103
ΔP <sub>B.A.</sub> "H <sub>2</sub> O	1.05	.75	.88	.83	.80
W <sub>B.A.</sub> lb/sec	2.16	2.06	1.48	1.42	1.89
P <sub>3</sub> "H <sub>2</sub> O	7.1	7.8	8.4	8.6	9.1
P <sub>T3</sub> "H <sub>2</sub> O	8.5	8.0	8.6	8.1	7.3
P <sub>4</sub> "H <sub>2</sub> O	.8	.55	.4	.32	.30
P <sub>t4</sub> "H <sub>2</sub> O	5.8	5.3	6.55	6.45	7.35
Δ	5.3	5.25	6.15	6.00	7.05
M <sub>t</sub>	.417	.500	.557	.545	.60

TEMP'S : COOLING AIR 80°F PRESS : ATMOSPHERIC 27.82 "Hg  
 AMBIENT TEST CELL 120°F AMBIENT TEST CELL 27.50 "Hg





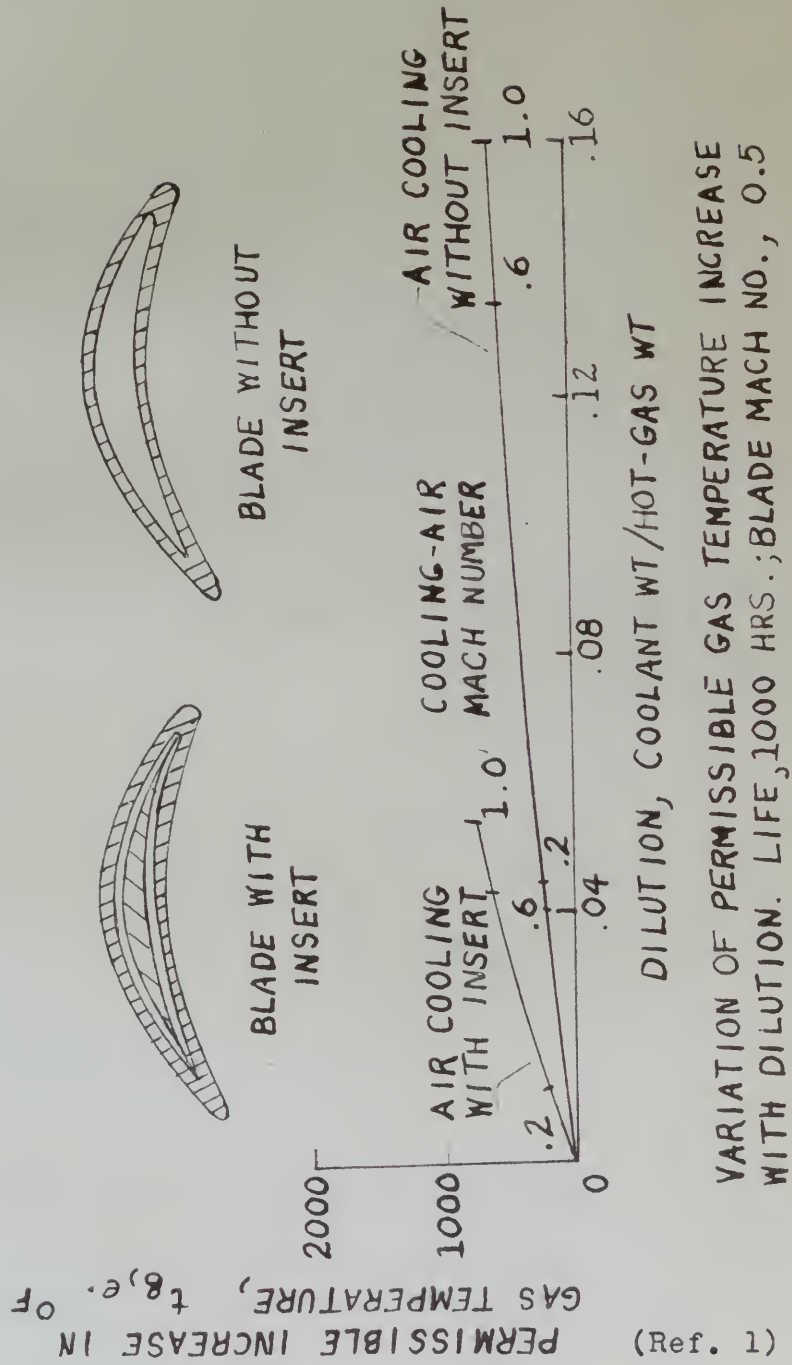


TURBOPROP ENGINE PERFORMANCE WITH +WITHOUT COOLING  
LOSSES. AIRPLANE SPEED, 500 MPH: MACH  
NO., 0.69; ALTITUDE 30,000 FT. (Ref. 1)

Fig. 1





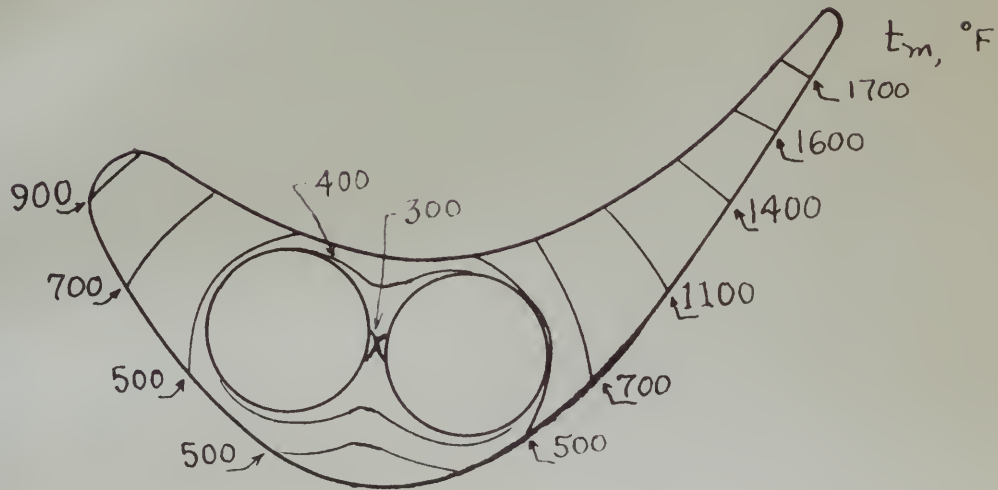


(Ref. 1)

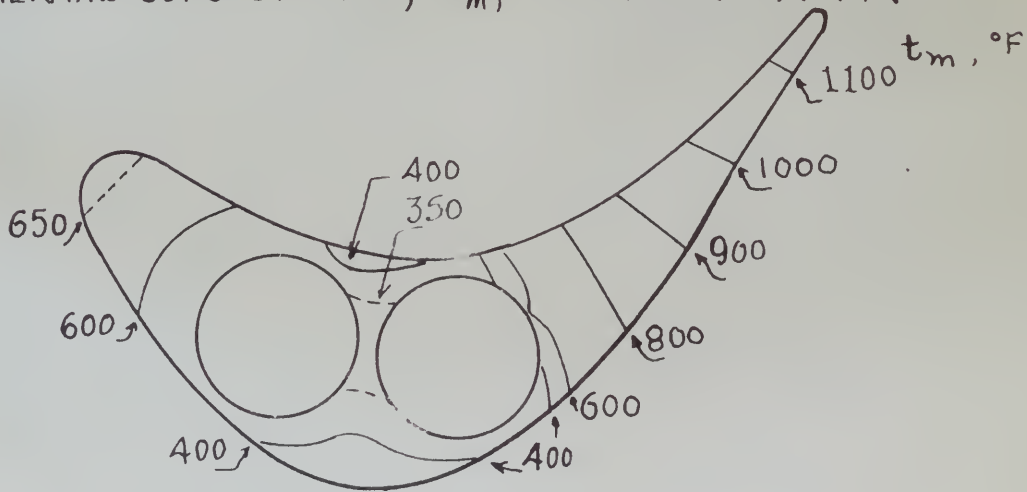
Fig. 2



THERMAL CONDUCTIVITY,  $k_m$ , 15 BTU/(HR)(°F)(FT)



THERMAL CONDUCTIVITY,  $k_m$ , 100 BTU/(HR)(°F)(FT)



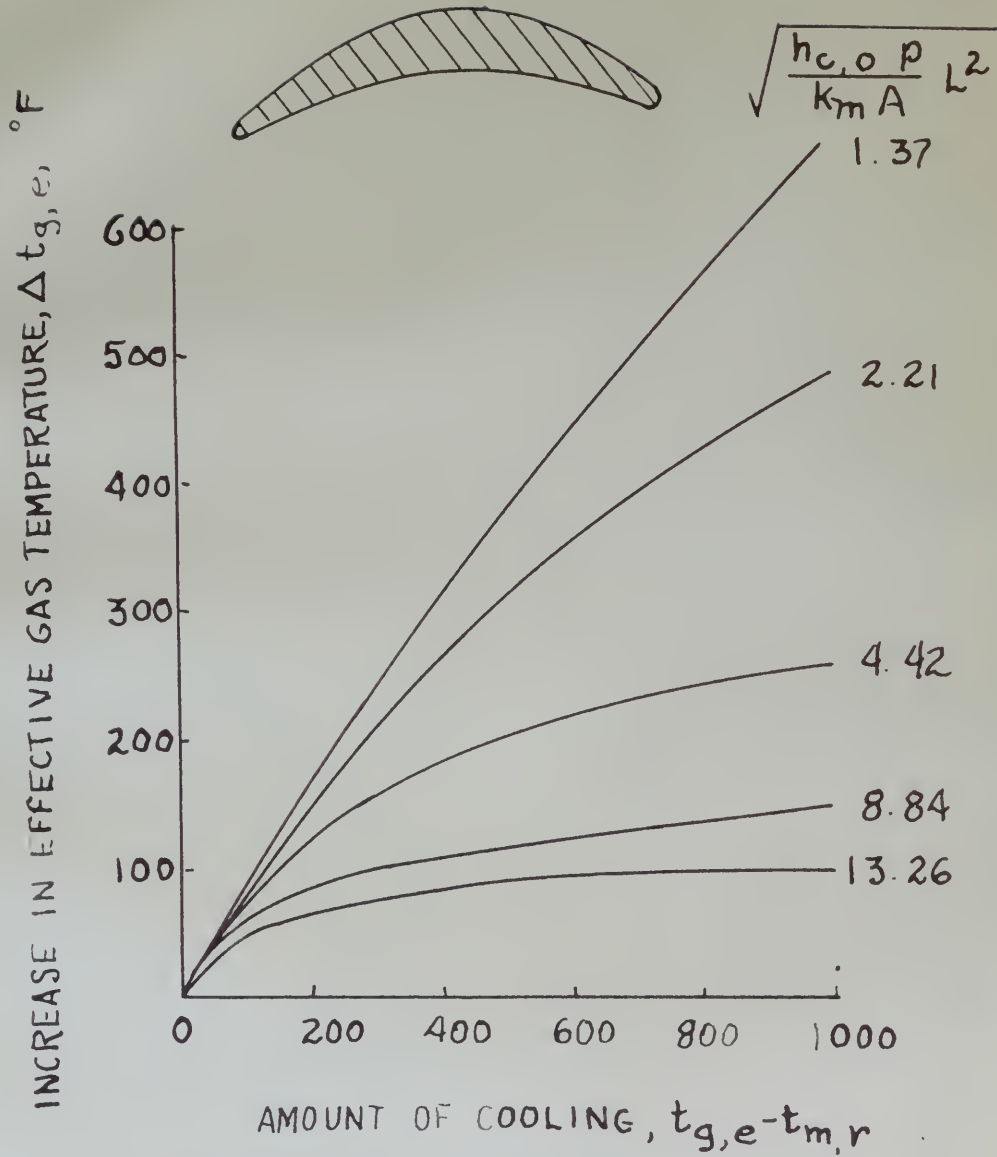
ISOTHERMS IN BLADE SECTIONS OF DIFFERENT CONDUCTIVITY MATERIAL WITH LIQUID COOLING. GAS FLOW, 55 LB/SEC; WATER FLOW, 6.42 LB/SEC; GAS TEMPERATURE, 2000° F; WATER TEMPERATURE, 200° F.

(Ref. 1)

Fig. 3





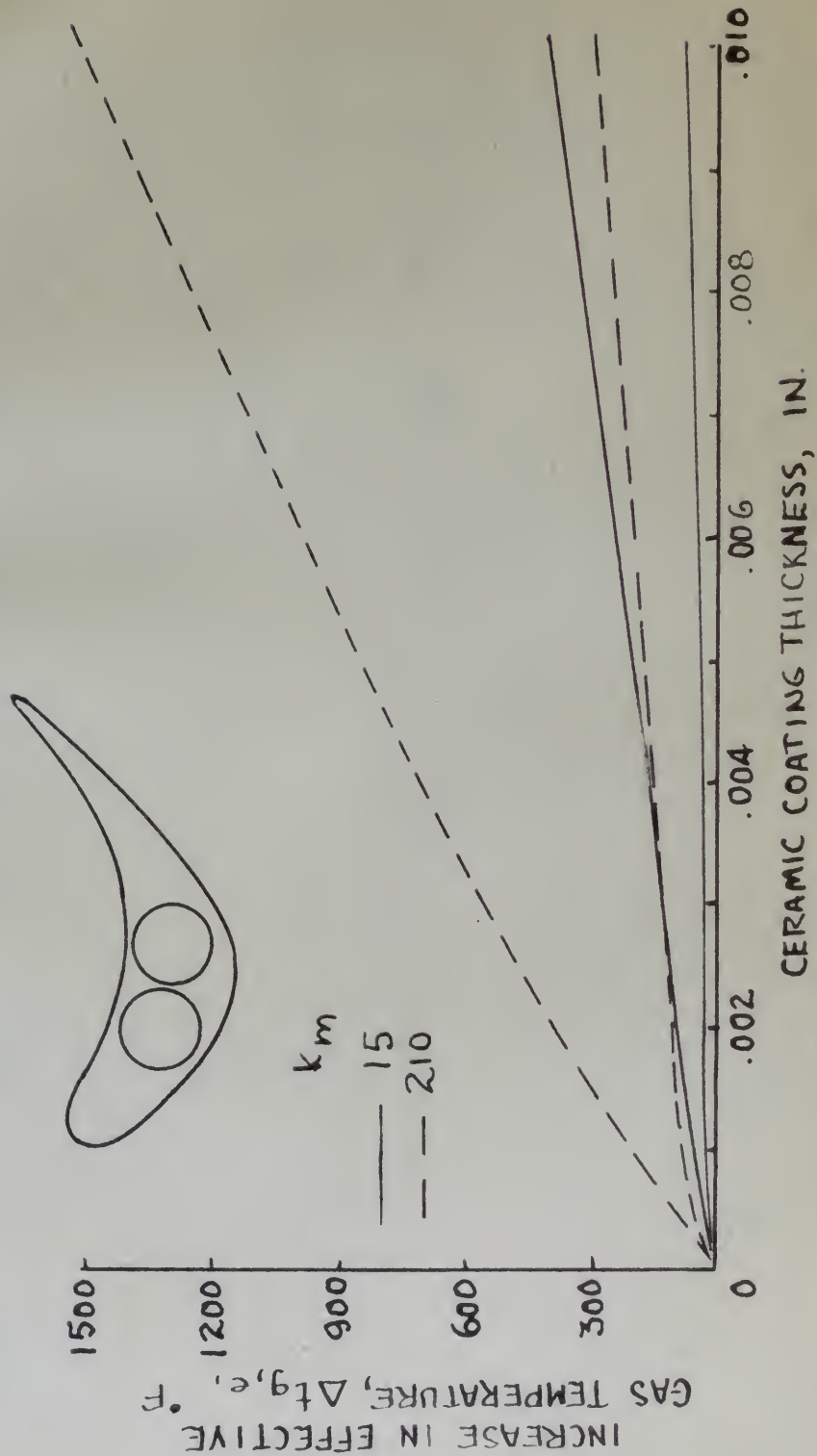


VARIATION OF RIM COOLING EFFECTIVENESS.  
 MAXIMUM ALLOWABLE MACH NUMBER, 0.5.  
 (Ref. 1)

Fig. 4







(Ref. 1)

Fig. 5

VARIATION OF INCREASE IN EFFECTIVE GAS TEMPERATURE WITH CERAMIC COATING THICKNESS FOR TWO METAL AND CERAMIC THERMAL CONDUCTIVITIES.



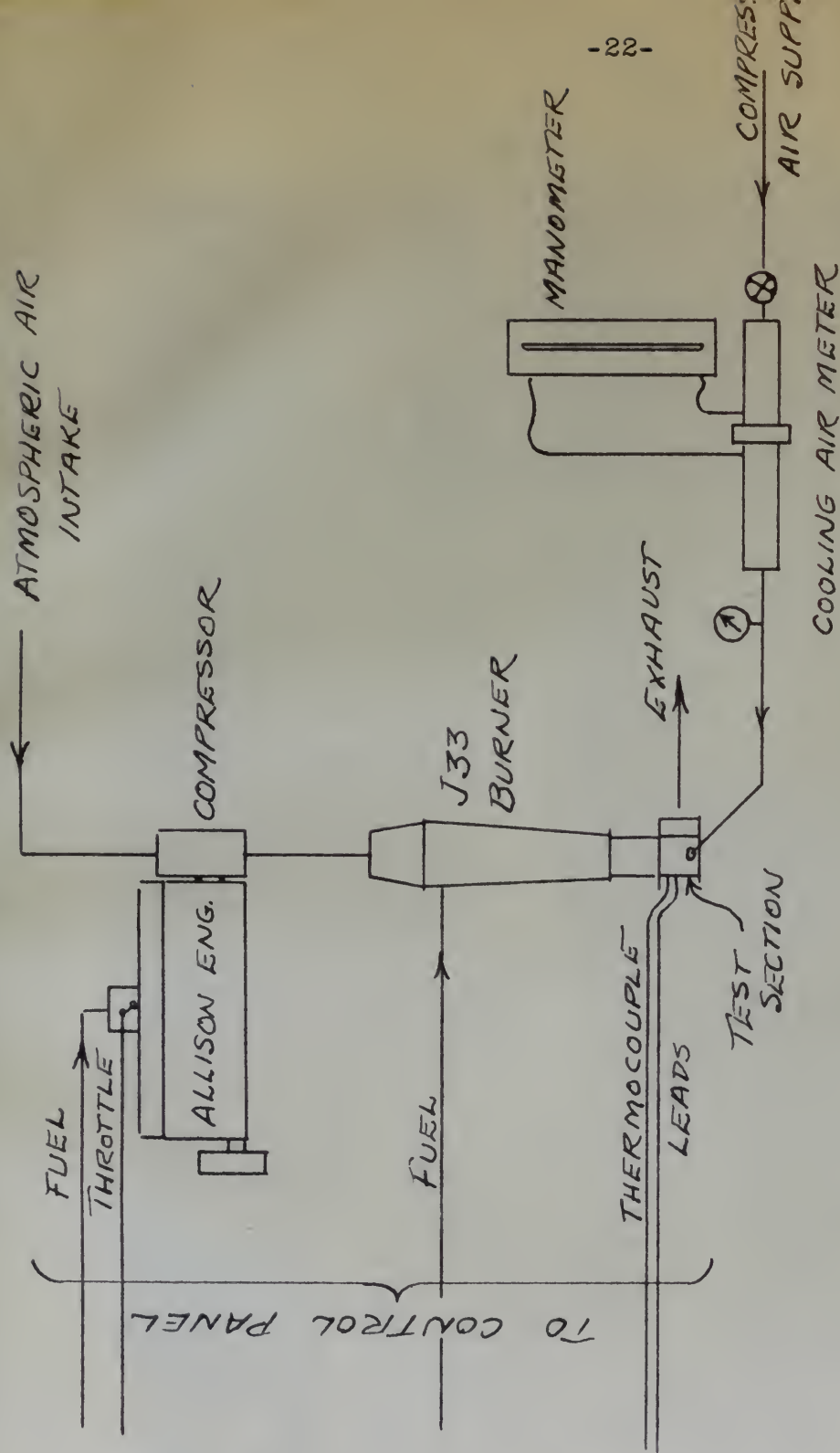
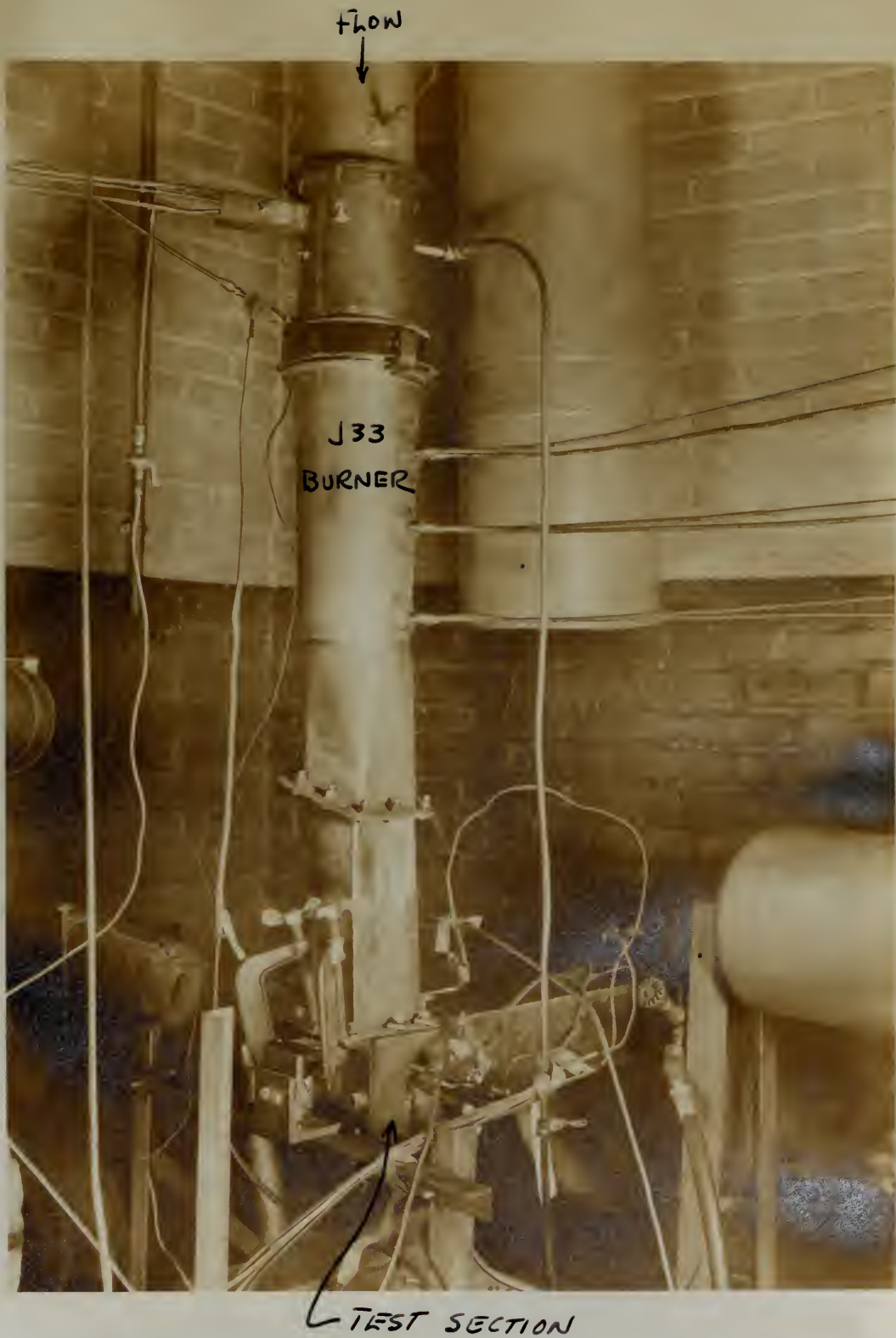


Fig. 6

SCHEMATIC DIAGRAM OF COMPLETE TEST LAYOUT

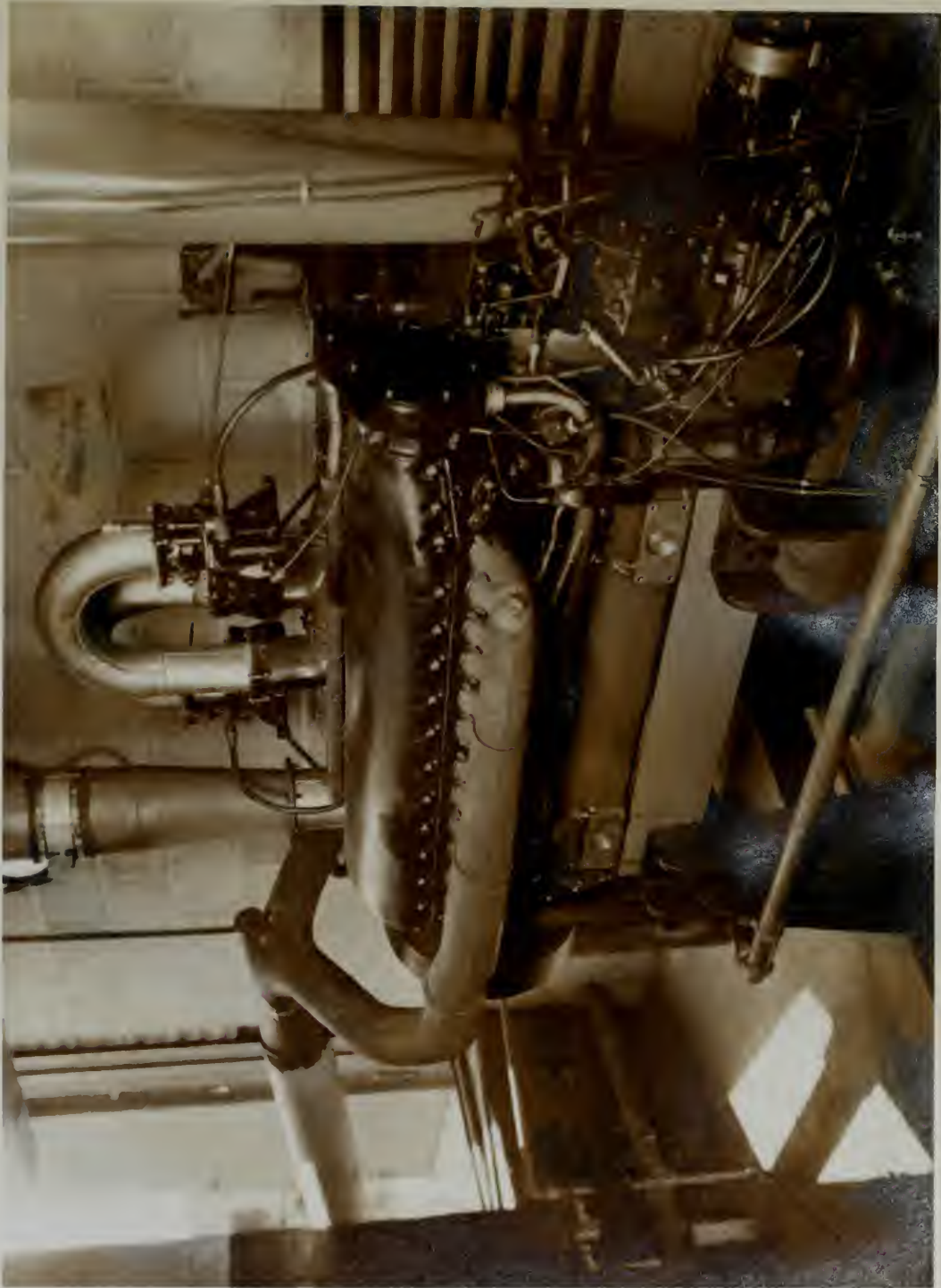






TEST SECTION MOUNTED ON BURNER



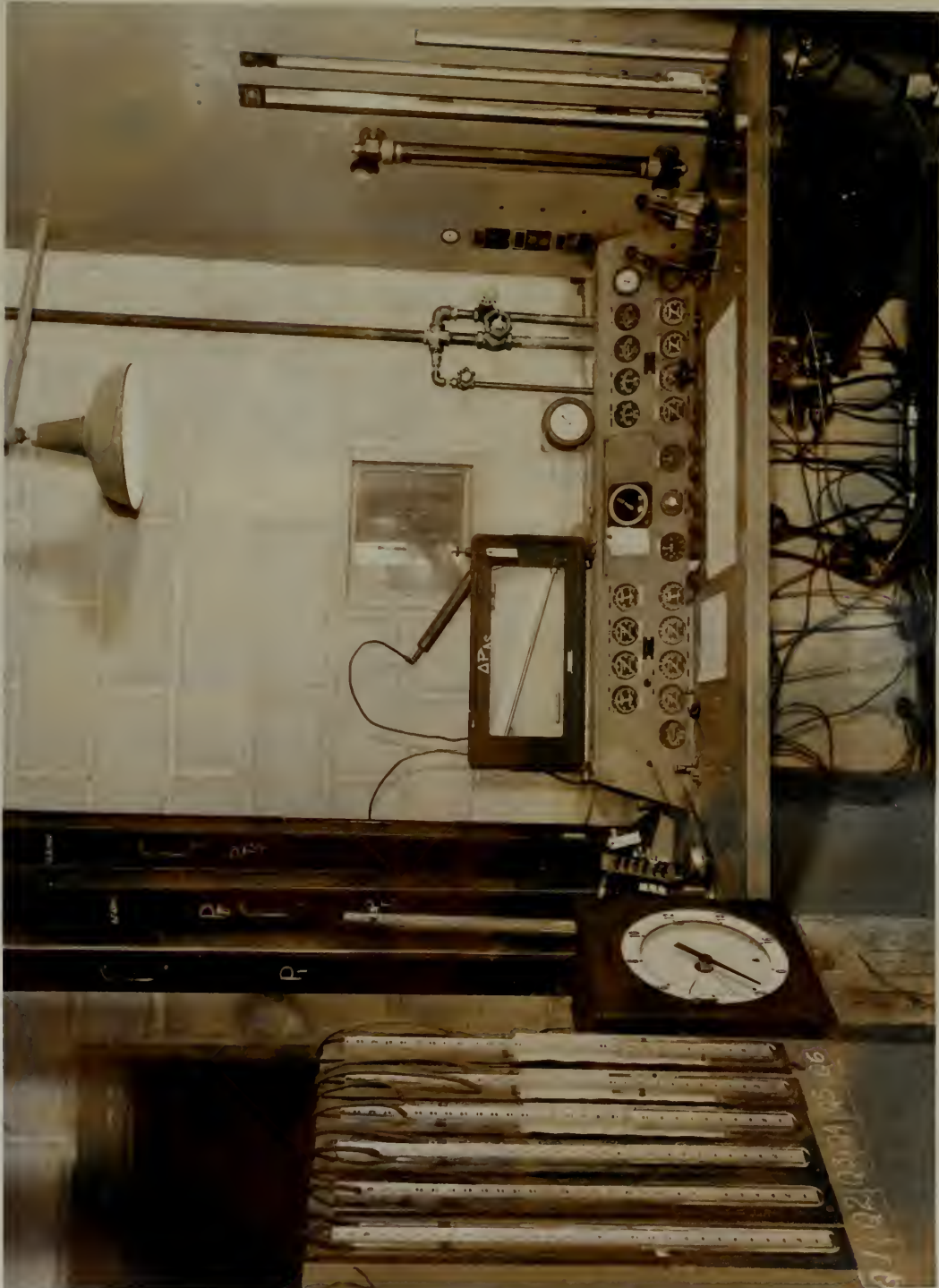


ANKER ENGINE

Fig. 8







CONTROL PANEL

Fig. 9





COOKING AIR METER

Fig. 10

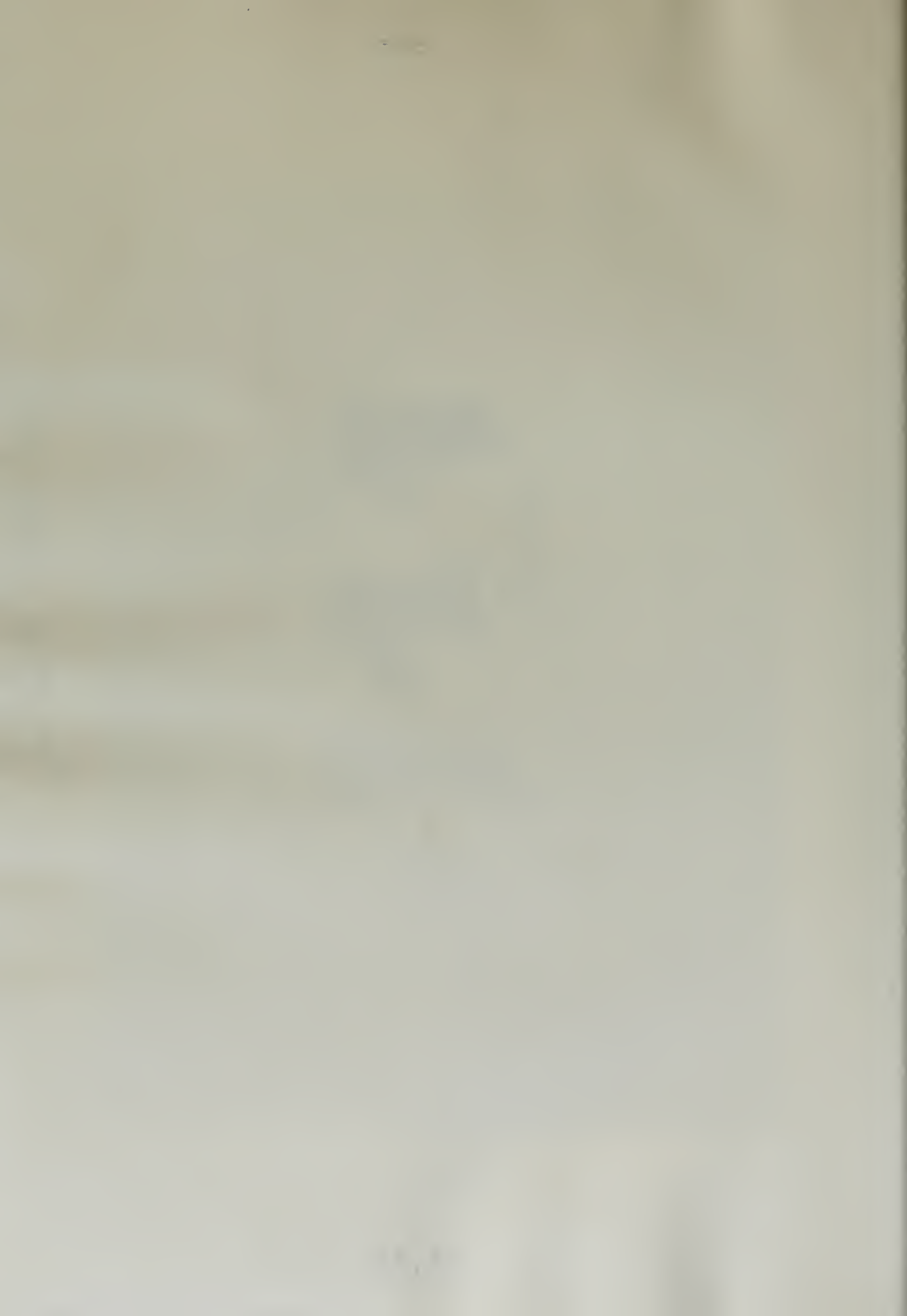




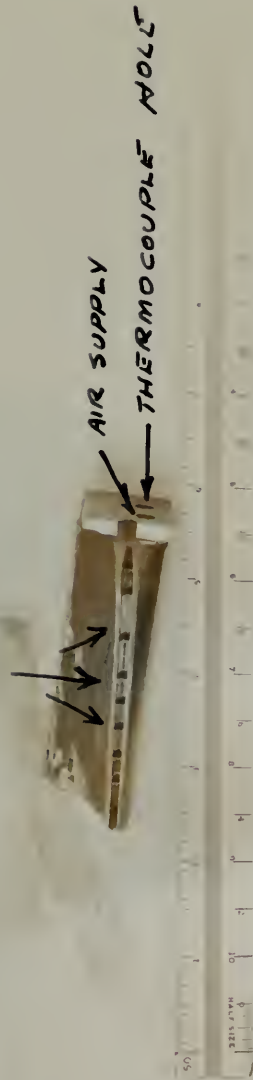


TEST BLADE (CONF. A)

Fig. 11



LEADING EDGE BLEEDS

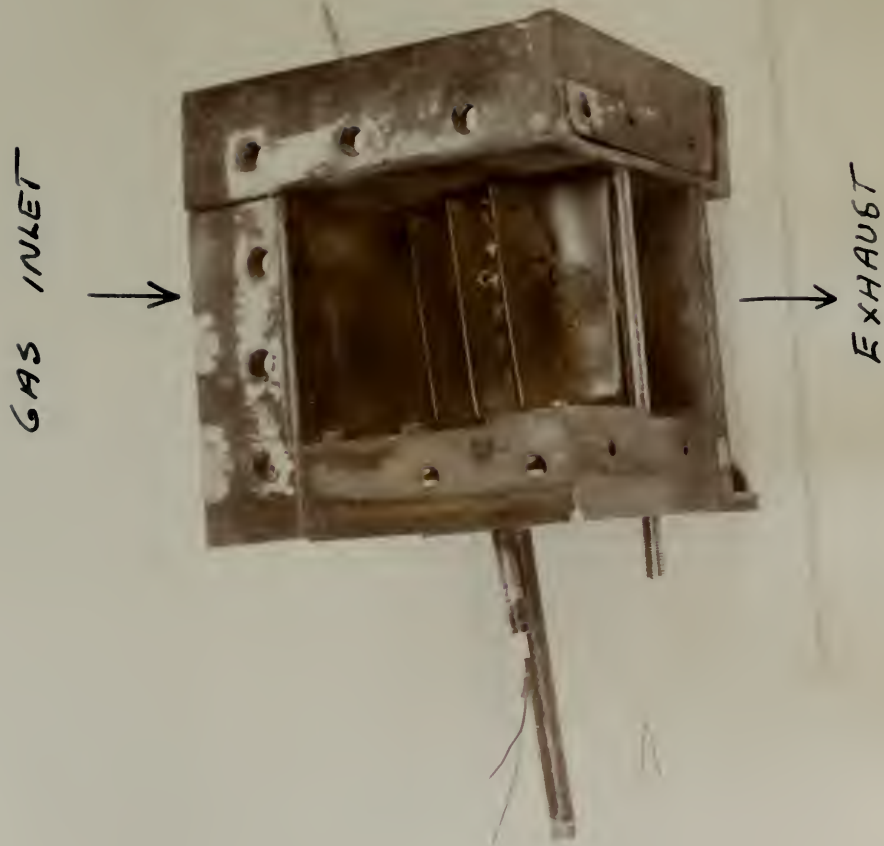


TEST BLADE (CONF. B)

Fig. 12







COMPLETE TEST SECTION

Fig. 13



- A- LEADING EDGE  
THERMOCOUPLE
- B- TRAILING EDGE  
THERMOCOUPLE
- C- COOLING AIR SUPPLY
- D- " " BLEEDS

-30-



MOUNTED TEST BLADE

Fig. 14





GAS INLET



EXHAUST



END VIEW OF BLADES

Fig. 15





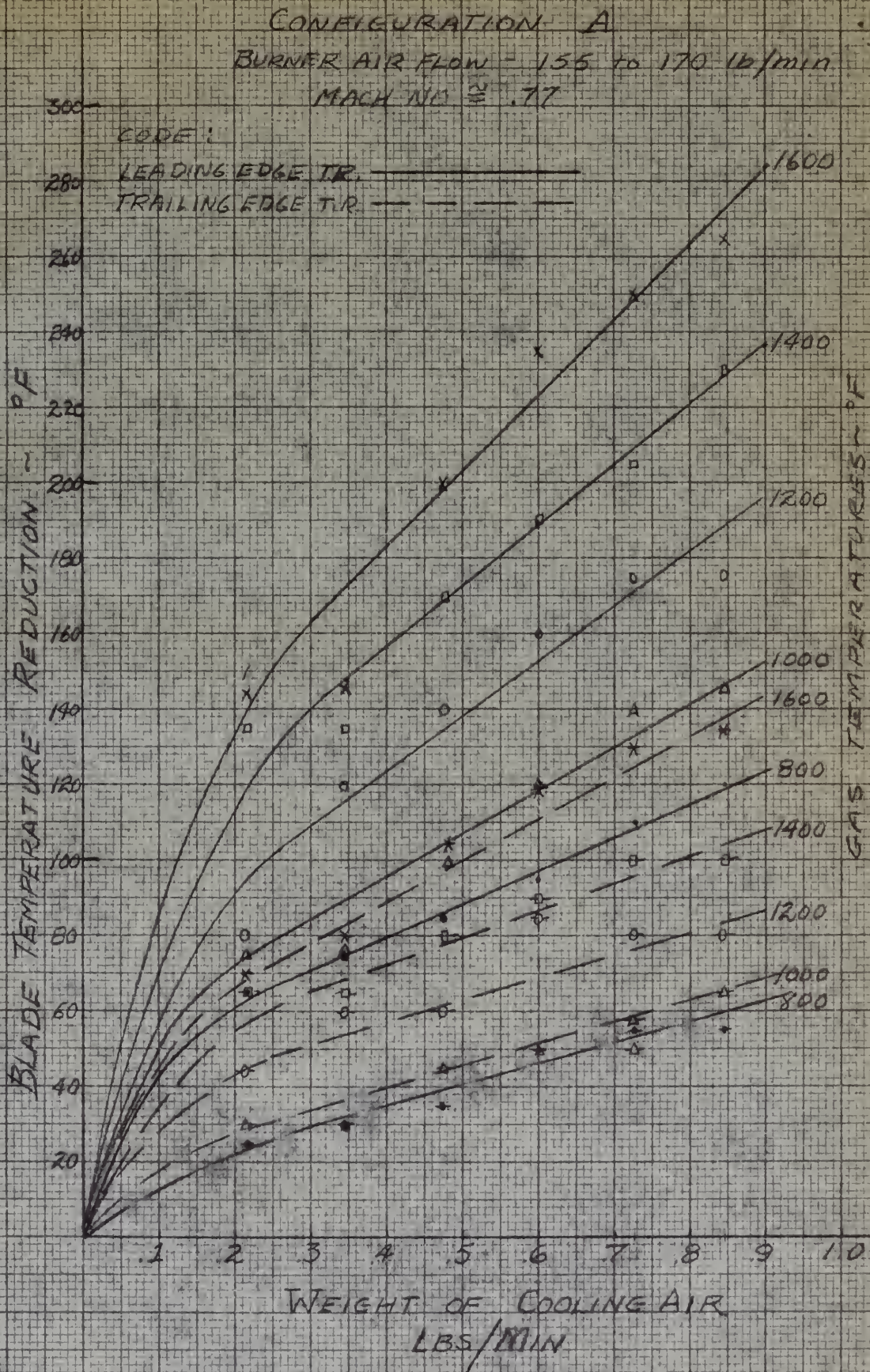
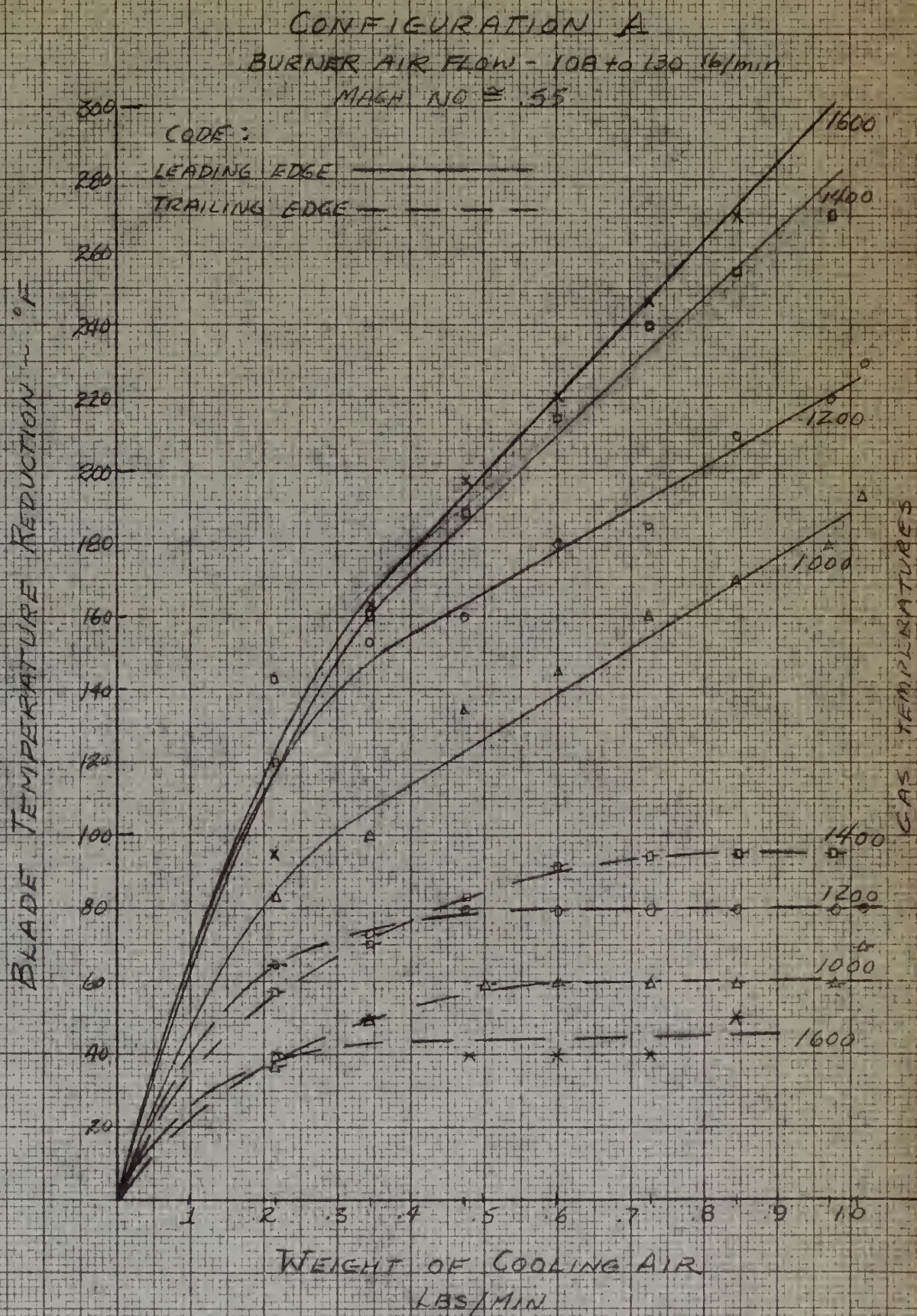


FIG 16





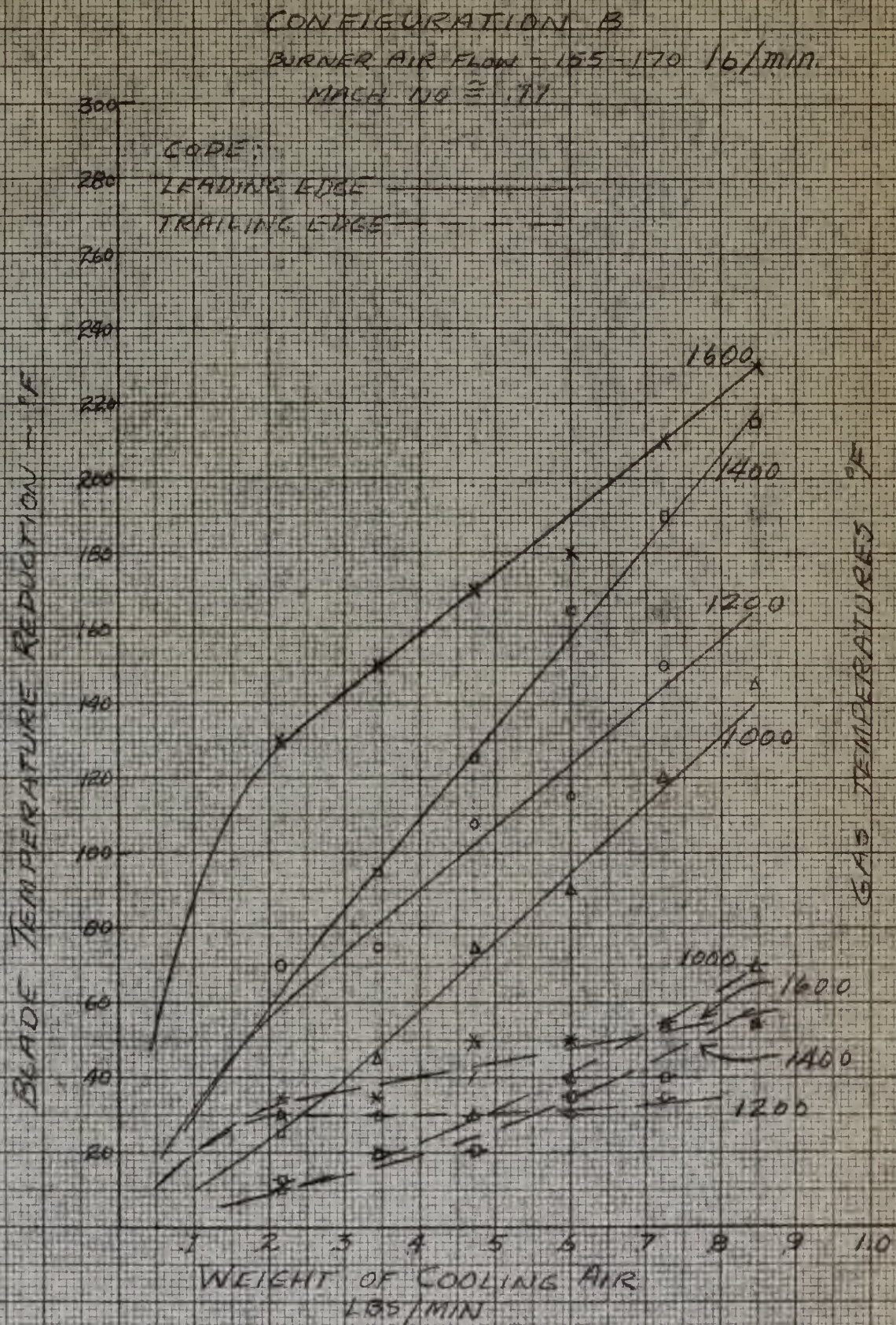




EFFECTIVENESS OF BOUNDARY LAYER  
 IN REDUCING BLADE TEMPERATURE







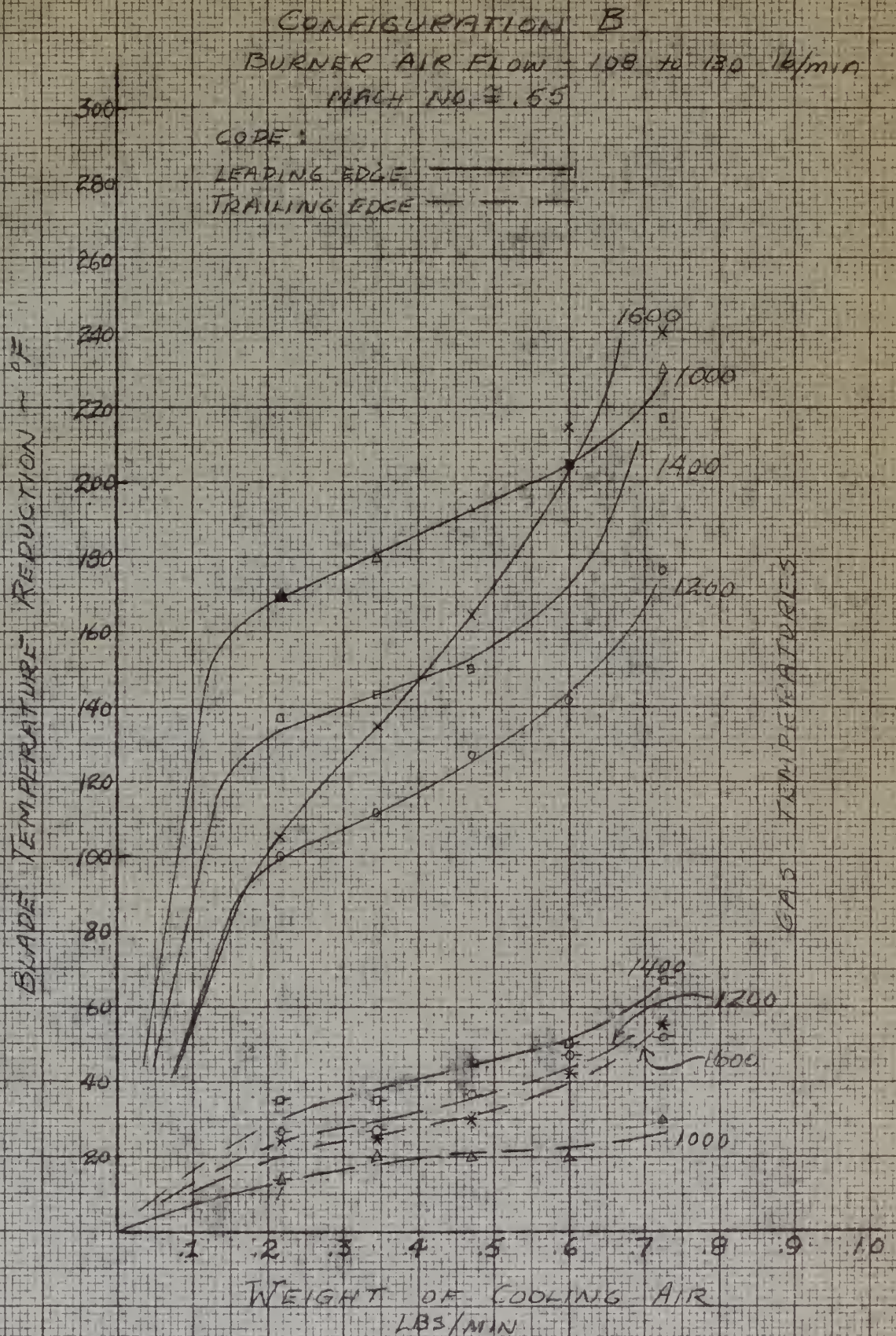
EFFECTIVENESS OF BOUNDARY LAYER  
IN REDUCING BLADE TEMPERATURES

Fig. 18



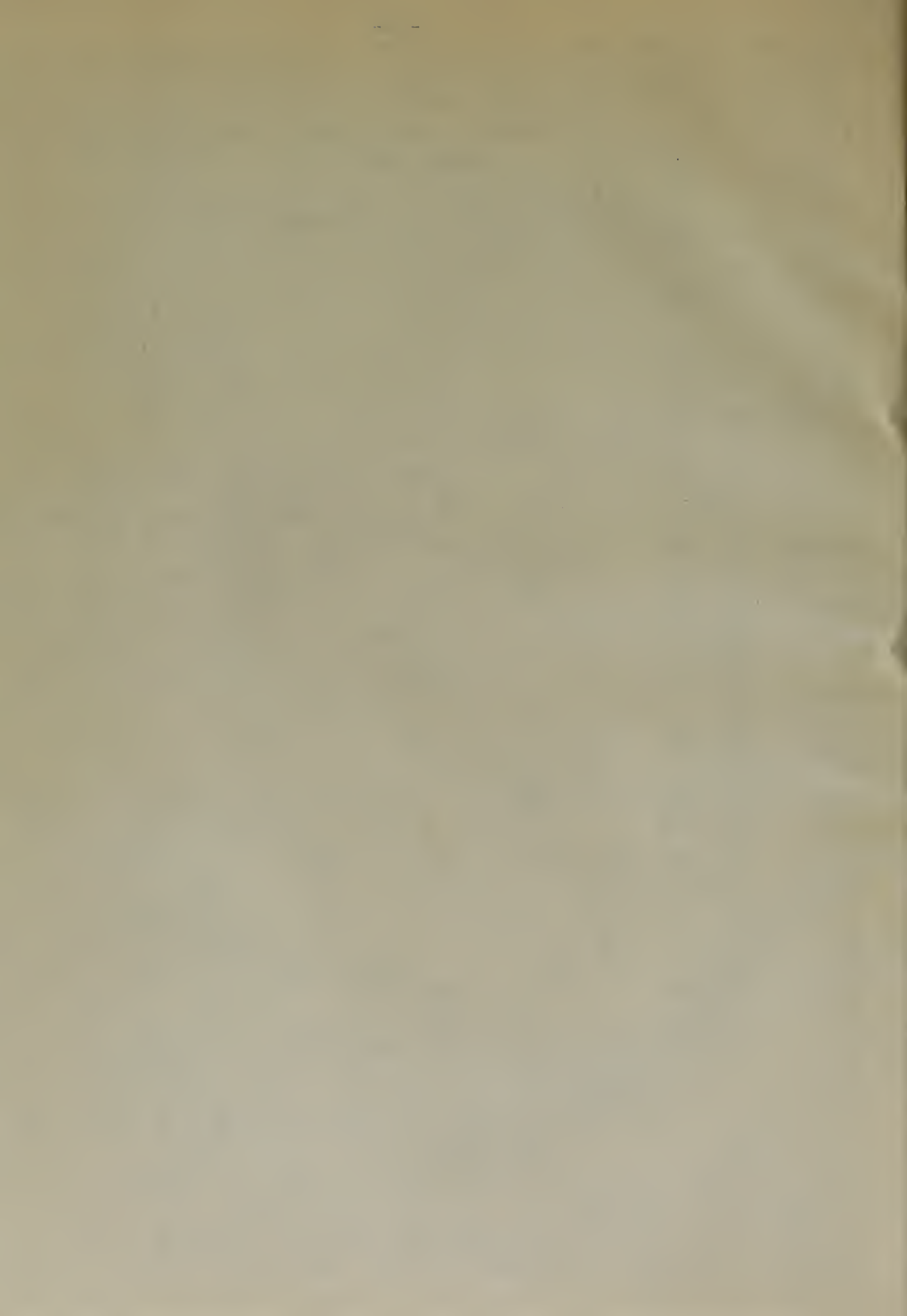






EFFECTIVENESS OF BOUNDARY LAYER  
 IN REDUCING BLADE TEMPERATURES

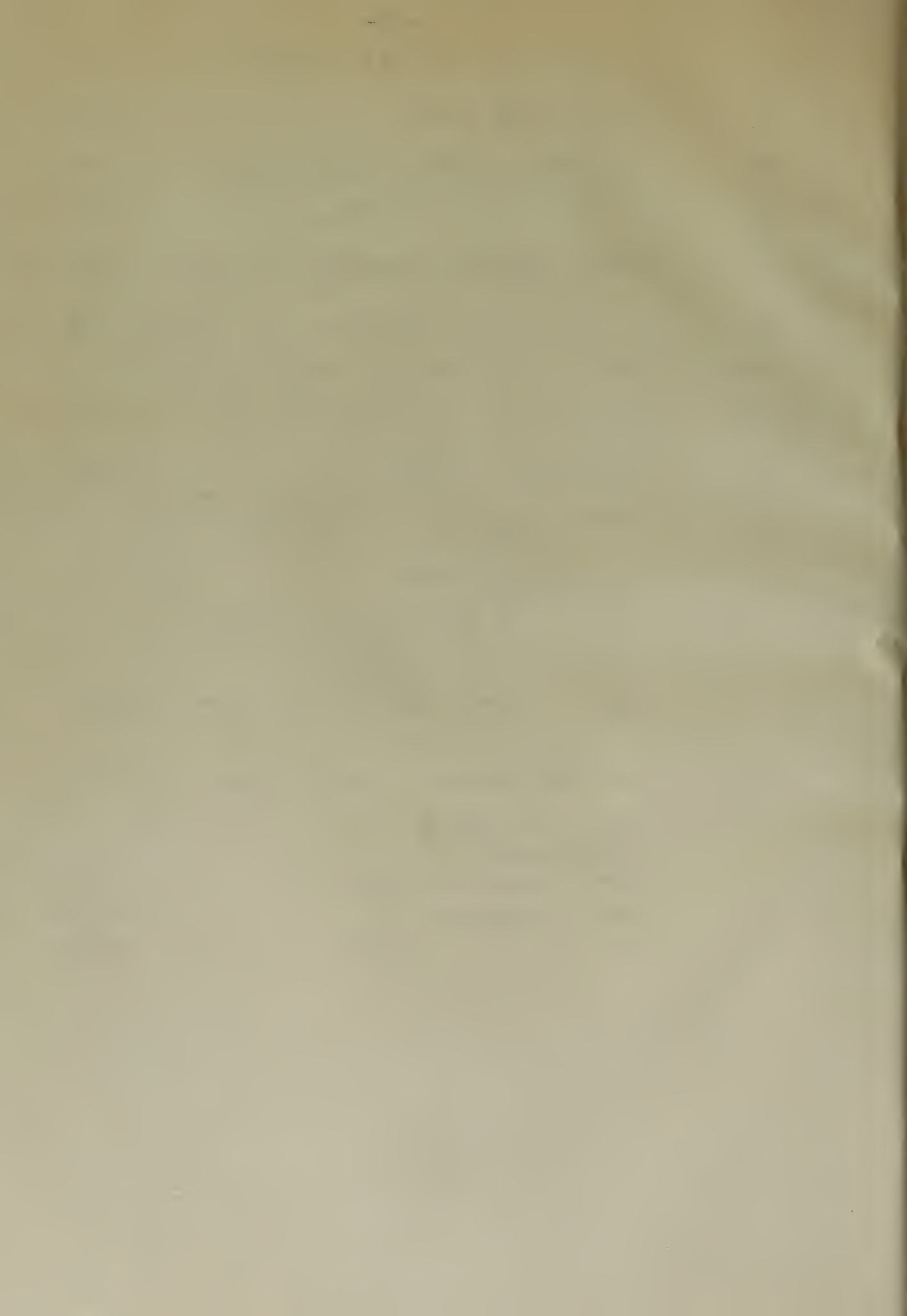
Fig. 19





# NOMENCLATURE

$P_t$	FUEL PRESSURE	psia
$P_3$	BURNER INLET STATIC PRESS	"Hg
$P_{t3}$	" " TOTAL "	"Hg
$P_4$	TEST SECTION STATIC "	"Hg
$P_{t4}$	" " TOTAL "	"Hg
$\Delta P_{CA}$	COOLING AIR ORIFICE PRESS DROP	"H <sub>2</sub> O
$\Delta P_{BA}$	BURNER " " "	"Hg
$q$	DYNAMIC PRESS	"Hg
$M_4$	TEST SECTION MACH NO.	
$T$	TEMPERATURE	°F
SUBSCRIPTS:		
	BLE BLADE LEADING EDGE	
	BTE " TRAILING "	
	M, R " ROOT	
	G, C GAS, AFFECTING HEAT TRANSFER	
T.R.	TEMPERATURE REDUCTION	°F
$W$	WEIGHT FLOW	
SUBSCRIPTS:		
	CA COOLING AIR	lb/min
	BA BURNER "	lb/sec
	f " FUEL	lb/hr





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- (4) McAdams, W. H.; "Heat Transmission;" 2nd Ed., McGraw-Hill Book Co., 1942.

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|-----|--|
| (1) | Winters, W. H. "The Development of the<br>"State System" - 1st ed. 1911. |
| (2) | Winters, W. H. "The Development of the<br>"State System" - 2nd ed. 1911. |
| (3) | Winters, W. H. "The Development of the<br>"State System" - 3rd ed. 1911. |
| (4) | Winters, W. H. "The Development of the<br>"State System" - 4th ed. 1911. |

# SAMPLE CALCULATIONS

Air metering: With standard sharp edged orifice configuration with flange taps.

Cooling air,

$$W_a = .8595 K D_2^2 \frac{(P_2 \times \Delta P)^{\frac{1}{2}}}{(T_a)^{\frac{1}{2}}} \quad (\text{ASME Power Test Code})$$

$T_a = 80^\circ \text{ F.}$  Air supply temperature

$D_2 = .75"$  Orifice diameter

$D_1 = 2.07"$  Pipe diameter

$K = .61$  From Fig. 34(a) of ASME Power Test Codes

$P_2 =$  Absolute outlet static pressure lb./in.<sup>2</sup>

$\Delta P =$  Orifice static pressure drop lb./in.<sup>2</sup>

$$W_a = \frac{.8596 \times .61 \times (.75)^2}{(540)^{\frac{1}{2}}} (P_2 \times \Delta P)^{\frac{1}{2}}$$

$$= .01268 (P_2 \times \Delta P)^{\frac{1}{2}}$$

P	P(psi)	P <sub>2</sub>	W <sub>a</sub> (lb./sec.)	W <sub>a</sub> (lb./min.)
.1	.0036	22.6	.00362	.217
.2	.0072	28.6	.00575	.345
.3	.0108	35.6	.00785	.471
.4	.0144	43.6	.01	.60
.5	.0180	50.6	.0121	.726
.6	.0216	57.6	.0141	.846
.7	.0252	64.6	.0162	.972
.8	.0288	71.6	.0182	1.09

Burner air,

$$W_{BA} = .8596 K D_2^2 \frac{(P_2 \times \Delta P)^{\frac{1}{2}}}{T_1^{\frac{1}{2}}}$$

$K = .704$

$D_2 = 5.6$

Results are tabulated in Tables I and II.

SAFETY CALCULATIONS

With standard sharp edged orifice  
ventilation with linear edge.

Flowing air.

$$Q = \frac{C_d A \sqrt{2 \Delta P}}{\rho}$$

(From Figure 10.1)

$$Q = 1.0 \text{ m}^3/\text{s}$$

air supply requirement

$$Q = 1.0 \text{ m}^3/\text{s}$$

system demand

$$Q = 1.0 \text{ m}^3/\text{s}$$

line demand

$$Q = 1.0 \text{ m}^3/\text{s}$$

from Fig. 10.1 at 1000 Pa static pressure

$$Q = 1.0 \text{ m}^3/\text{s}$$

through orifice static pressure

$$\Delta P = 1.0 \text{ Pa}$$

orifice static pressure drop

$$Q = 1.0 \text{ m}^3/\text{s}$$

load

$$Q = 1.0 \text{ m}^3/\text{s}$$

Q (m³/s)	P (Pa)	Q (m³/s)	P (Pa)
1.0	1000	1.0	1000
0.8	640	0.8	640
0.6	360	0.6	360
0.4	160	0.4	160
0.2	40	0.2	40
0.1	10	0.1	10
0.05	4	0.05	4
0.02	1	0.02	1
0.01	0.25	0.01	0.25
0.005	0.06	0.005	0.06
0.002	0.01	0.002	0.01
0.001	0.0025	0.001	0.0025

Flowing air.

$$Q = 1.0 \text{ m}^3/\text{s}$$

$$Q = 1.0 \text{ m}^3/\text{s}$$

Results are tabulated in Tables 1 and 2.



Mach. number:

$$q = \frac{\gamma}{2} \rho V^2$$

$M_4$  = Test section mach. no.

$q$  = Test section dynamic pressure

$\gamma = 1.3$  for gas

$P_4$  = Test section static pressure

$$M_4 = \frac{2q}{\gamma P_4} = \frac{q}{.65P_4}$$

Results are tabulated in Tables I and II.





**DATE DUE**

27 MR '5

[illegible]



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Boundary layer control  
as a method of gas tur-  
bine blade cooling.

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